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Analysis of solar desiccant cooling system for an institutional building in subtropical Queensland, Australia

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A B S T R A C T

Institutional buildings contain different types of functional spaces which require different types of heating, ventilating and air conditioning (HVAC) systems. In addition, institutional buildings should be designed to maintain an optimal indoor comfort condition with minimal energy consumption and minimal negative environmental impact. Recently there has been a significant interest in implementing desiccant cooling technologies within institutional buildings. Solar desiccant cooling systems are reliable in performance, environmentally friendly and capable of improving indoor air quality at a lower cost. In this study, a solar desiccant cooling system for an institutional building in subtropical Queensland (Australia) is assessed using TRNSYS 16 software. This system has been designed and installed at the Rockhampton campus of Central Queensland University. The system's technical performance, economic analysis, energy savings, and avoided gas emission are quantified in reference to a conventional HVAC system under the influence of Rockhampton's typical meteorological year. The technical and economic parameters that are used to assess the system's viability are: coefficient of performance (*COP*), solar fraction, life cycle analysis, payback period, present worth factor and the avoided gas emission. Results showed that, the installed cooling system at Central Queensland University which consists of 10 m^2 of solar collectors and a 0.400 m^3 of hot water storage tank, achieved a 0.7 *COP* and 22% of solar fraction during the cooling season. These values can be boosted to 1.2 *COP* and 69% respectively if 20 m^2 of evacuated tube collector's area and 1.5 m^3 of solar hot water storage volume are installed.

Contents

1. Introduction

Institutional buildings contain different types of functional spaces. Lecture theatres, libraries and laboratories are the most important spaces within institutional buildings and they are

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usually the largest air conditioned area which host the daily occupants' activity, machinery and instruments. In institutional buildings, HVAC is a very important means to maintain a comfortable living space and to provide clean air to the occupants. However, the supplied cooled air is often contaminated with dust, microbes, viruses and fungi [1]. In addition, high indoor humidity is a major contributor to the accumulation of moisture in a buildings envelope. These often cause dampness within the building and subsequent health-related problems for the occupants. Moreover indoor humidity affects humans, library contents (books and furniture) and laboratories (machines and equipment). Institutional buildings have a very high occupational density compared to most other commercial buildings. This high occupancy density generates a high heat gain as well as a high emission of body odours and water vapour. It is known that the human body has a constant temperature of 36–37 °C, independent of surrounding conditions and muscle activities. As a consequence, the human body has to transmit the excess heat to the environment by means of different heat transfer mechanism. This excess heat consists of latent and sensible heat. The sensible heat is transferred by means of convection and radiation from the human body to its surroundings, while latent heat is transferred to the surrounding by diffusion of vapour through skin and exhaled air [2]. The most common practice to overcome contaminations and to ensure an acceptable air quality is using hypo filters in conjunction with dampers. Conversely, the hypo filters which are made of organic material act as a nidus for the growth of fungi in the presence of HVAC condensing moisture [3]. While the most common procedure to conquer the HVAC's condensing water issue is by overcooling the treated air below the dew point this is also considered an expensive practice. Designers, developers and architects are therefore urged to use non conventional HVAC system's as solar cooling technologies. Furthermore, using non conventional HVAC technologies which use clean materials and renewable energy resources can significantly reduce building energy consumption and enhance indoor air quality. In solar cooling systems, solar heat is required to drive the cooling process, and this can be done by collecting solar radiation using solar collectors to convert it into thermal energy, and then this energy is used to drive thermally driven cooling cycles such as desiccant, absorption and adsorption cycles [4].

Solar desiccant cooling systems are an energy efficient and environmentally friendly way to improve indoor air quality due to its superior latent load control. This technology is considered as a great alternative for conventional air conditioning in commercial buildings particularly institutional buildings and health care buildings to reduce contaminated air transmissions [5]. The solar desiccant cooling systems works as an open cycle which is based on the combination of desiccant processes and evaporative cooling. The main principle of the desiccant cooling cycle is the systems capability of removing or reducing vapours and moisture out of the treated air using a physical sorption process [6]. The sorption process can be undertaken using adsorption or absorption. The adsorption process is a physical process where the property of the desiccant material remains unchanged, while in the absorption process, the physical characteristic of the material changes when attracting moisture [6]. Desiccant materials are available as solid or liquid. Example of desiccant materials are silica gel, titanium silicates, calcium chloride, activated aluminas, zeolite (natural and synthetic), molecular sieves, lithium chloride, organic-based desiccants, polymers, compound and composite desiccants [7]. Market available desiccant systems include liquid spray towers, solid packed tower, rotating horizontal bed, multiple vertical bed and rotating desiccant wheel [8]. The main component of a solar desiccant cooling cycle are the solar energy system which consists of solar collectors and hot water storage, the dehumidifier which consists of a rotating wheel containing desiccant material and the evaporative cooler. The cooling process starts in the dehumidifier as shown in Fig. 1, where the desiccant material dries the supplied air to produce a dry and warm air. Then the warm and dry air is passed to the evaporative cooler to reduce its sensible temperature to near ambient conditions and then to the cooled space. This continuous air drying eventually makes the desiccant materials saturated, and cannot function again unless regenerated. In order to use the desiccant material again, thermal energy is required for the regeneration process. Generally this thermal energy can be supplied by gas or solar [6,9].

Solar desiccant cooling systems can deliver a dryness enough to treat 7.5 l of wet air per second per person and the personal moisture load of 70 W latent (0.1 l per hour) [10,11]. Due to using a low grade thermal energy and environmentally friendly materials, solar desiccant cooling systems have become attractive to researcher's to resolve issues associated with using conventional air conditioning and indoor air quality.

A number of studies have been carried out to investigate and evaluate solar desiccant cooling technology. Experimental investigations started when Baum et al. [12] followed by Kakabaev et al. [13] investigated a liquid desiccant cooling system based on a theory presented by Kakabaev and Khandurdyev [14,15]. Dai et al. examined the numerical performance of a hybrid solar solid desiccant cooling system for grain storage [16]. Halliday et al. demonstrated the potential of using solar energy to drive a desiccant cooling system [17]. Goldsworthy and White have recently investigated a desiccant cooling system with an indirect evaporative cooler [18]. Alizadeh performed an experimental study of a forced flow solar collector regenerator using liquid desiccant for Brisbane climate [19]. Ismail et al. evaluated a solar regenerated open cycle desiccant bed system used in a grain storage in Melbourne [20]. Leutz et al. have investigated sorption

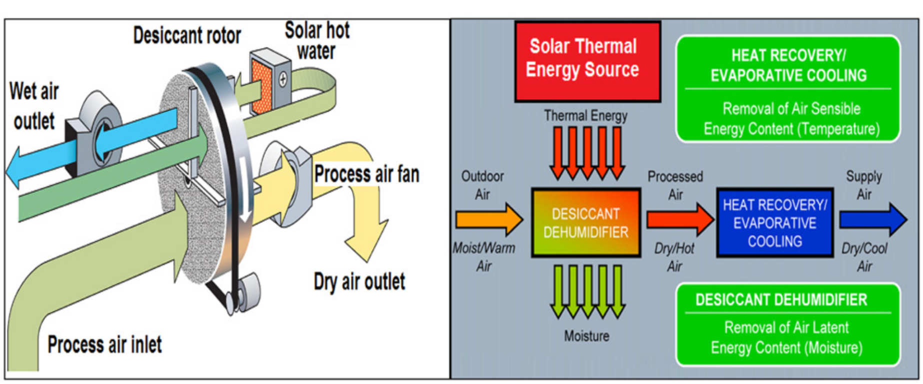

Fig. 1. Desiccant cycle and the operational concept of desiccant cooling system.

cycles for air conditioning applications for climates prevalent in Australasia [21]. White et al. have investigated performance of solar desiccant cooling system in the warm temperature climates of Melbourne, Sydney and the tropical climate of Darwin [22]. Most of the research and publications concerned with energy performance and indoor air quality of institutional buildings have only considered larger Australian cities. There are no such studies in regards to solar cooling technologies that have been under taken in Australian regional or Australian subtropical climates areas like Central Queensland. Considering the economic and environmental benefits of solar desiccant cooling technologies it is necessary to undertake a study on solar desiccant cooling system for various Australian climates.

In the present work an extensive numerical evaluation of a Central Queensland University solar desiccant cooling system is presented using TRNSYS software taking into account installing different solar collector's area and different hot water storage volume under Central Queensland subtropical climate. The main objective of this study is to assess the potential and the feasibility of the installed cooling system.

2. System technical analysis

The designed and installed solar desiccant cooling system at Central Queensland University, Rockhampton, consists of two subdivisions: the first subdivision is the solar energy system which consists of solar collectors and hot water storage and the second subdivision is a desiccant machine coupled with an evaporative cooler as shown in Fig. 2.

The efficiency of a desiccant cooling system can be evaluated based on its coefficient of performance (*COP*). Coefficient of performance (*COP*) is the ratio between the cooling capacity required to supply air conditioning (Q_C), and supply heat input needed for regeneration (Q_{Re}) as shown in Eq. (1) [23].

$$COP = \frac{Q_C}{Q_{Re}+Q_{evap}} = \frac{\eta_{heater} \times m_{sup} \times \Delta h_{cool}}{(m_{reg} \times \Delta h_{reg})+Q_{evap}} \tag{1}$$

where Q_{evap} is the energy consumed by the evaporative cooler (kW), η_{heater} is the regeneration heater efficiency, m_{sup} is the mass flow of supply air (kg/h), m_{reg} is the mass flow of the regeneration air kg/h, Δh_{cool} is the enthalpy difference between outside and supply air (kJ/kg), and Δh_{reg} is the enthalpy rise in the heater for the regeneration (kJ/kg).

When the mass flow of the supply air is the same as the regenerated air mass flow, heater efficiency ≈ 1, and the evaporative energy can be neglected compared to total energy, the *COP* of the desiccant cooling system can then be expressed as in Eq. (2).

$$COP = \frac{\Delta h_{cool}}{\Delta h_{reg}} \tag{2}$$

Thus using solar energy to regenerate the desiccant materials, the solar efficiency is given by [24].

$$\eta_{solar} = \eta_0 - C_1 \times \frac{t_m - t_a}{G} - C_2 \times \frac{(t_m - t_a)^2}{G} \tag{3}$$

Where η is the collector efficiency, η_0 is the optical efficiency, C_1 and C_2 are the collector heat loss coefficients, t_m is the collector temperature and t_a is the ambient temperature in °C.

Then the coefficient of performance for the solar desiccant cooling system is defined as Eq. (4).

$$COP = \frac{\Delta h_{cool}}{\Delta h_{reg}} \eta_{solar} = \left(\frac{\Delta h_{cool}}{\Delta h_{reg}}\right)\left(\eta_0 - C_1 \times \frac{t_m - t_a}{G} - C_2 \times \frac{(t_m - t_a)^2}{G}\right) \tag{4}$$

The thermodynamic measure of this system is based on the flow of energy between various system components. Normally in any solar cooling system the solar collectors convert solar radiation into thermal energy, and then a backup heater will be used if the solar thermal energy is insufficient to drive the cooling process. The total thermal energy generated from the solar collectors and the backup heater is known as the cooling system driving energy, and the percentage ratio of the thermal energy produced by the solar collectors to the total cooling system driving energy is known as the solar fraction (*SF*) which can be expressed as [25].

$$SF = \frac{QU}{Q_{total}} \tag{5}$$

Where *QU* is the thermal energy produced by the solar collectors and Q_{total} is the system total diving energy produced by the solar collectors and the backup heater.

The potential electric power saving E_{saved} for producing 1 kW cooling power is evaluated based on a comparison between conventional HVAC system and the solar desiccant cooling system as in Eq. (6) [23].

$$E_{Saved} = \frac{(W_{Conv}/Q_{C,Conv}) - (W_d/Q_{C,D})}{(E_{Conv}/Q_{C,Conv})} \tag{6}$$

Where W_{Conv} is the conventional system electric power in kW, $Q_{C,Conv}$ is the conventional cooling system capacity in kW, W_d is the desiccant cooling system electric power in kW and $Q_{C,D}$ is the desiccant cooling system cooling capacity in kW.

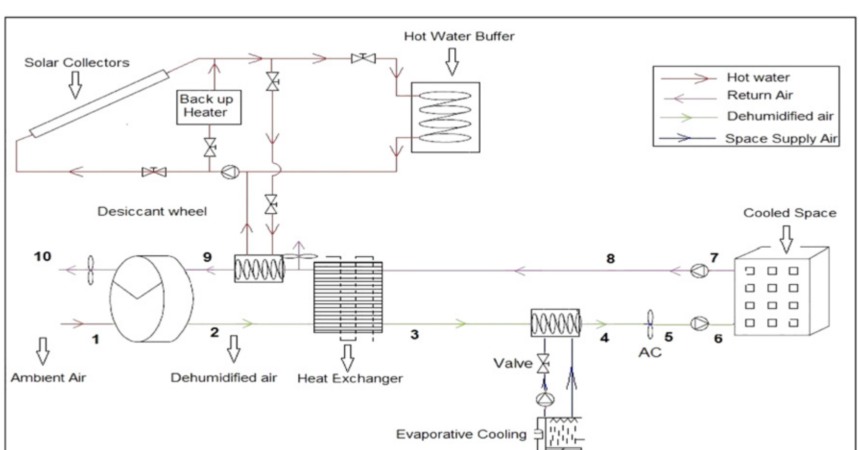

Fig. 2. Schematic diagram of the designed CQUniversity desiccant cooling system.

3. System economic analysis

Solar assisted air conditioning is characterised by high installation cost and cheap operating costs. Most solar cooling systems require a backup heat source; the heater is generally fitted as a part of the system. The total energy delivered by the backup heat source and the solar collectors is the total energy used by the system to cover the cooling load. The main objective behind solar process economical analysis is to determine the minimum cost of delivering the required solar energy. In other words for solar energy processes, the problem is to establish the right size of the solar cooling system that provide and guarantee the lowest cost combination of solar energy and conventional energy [24].

There are different methods that can be used to assess the economic performance of solar assisted air conditioners. The most used methods are present worth factor, life cycle analysis and payback period. Those methods are simple and practical to drive the system optimization in term of its cost, the cooling load and design parameters. In general, the cost of solar cooling process is linked to solar collector's area and solar fraction.

The installation costs of solar desiccant cooling equipments C_S consists of two terms: first term is proportional to collector's area and the other is independent of collector's area as shown in Eq. (7) [24].

$$C_S = (C_A \times A_C + C_E) \tag{7}$$

Where C_A is the cost of solar equipment's area, A_C is the solar collector's area and C_E is the equipment cost independent of collector's area.

Present worth factor is used to compare the future cost of a solar system to today's cost taken into account an obligation recurs each year at i inflation rate and d discount rate over N years or time period as in Eq. (8). Payback period is the total time that the system will cover its installation cost as expressed given by Eq. (9).

$$PWF\,(N,i,d) = \sum_{j=1}^{N} \frac{(1+i)^{j-1}}{(1+d)^j} = \left\{ \begin{array}{l} \frac{1}{d-i}\left[1-\left(\frac{1+i}{1+d}\right)^N\right] \rightarrow i \neq d \\ \frac{N}{d+1} \rightarrow i = d \end{array} \right\} \tag{8}$$

$$N_P = \frac{\ln\left[\frac{C_S \times i}{F \times L \times C_F} + 1\right]}{\ln(1+i)} \tag{9}$$

Life cycle savings of a solar desiccant cooling system compared to a conventional system can be defined as the difference between the savings in fuel cost and the increase of the expenses that occur as a result of solar system investment as given by Eq. (10).

$$LCS = P_1 \times C_F \times L \times F - P_2 \times C_S \tag{10}$$

Where C_F is the first period's unit energy cost delivered from fuel, L is the annual load, F is the annual solar fraction, P_1 is the factor relating to life cycle fuel cost savings in the first year and P_2 is the factor relating life cycle expenditures occurred by additional capital investment to the initial cost. P_1 and P_2 can be calculated as stated in [24].

In summary, the equations mentioned in this section were used to assess the solar system economics assuming the system is a non-income producer and future resale value is considered.

4. Environmental analysis

The state of Queensland, Australia, faces a high growth rate of population and urbanisation, which increases the rate of energy consumption. The reliance on fossil fuel to generate energy has contributed to Australia having the highest green house gas emissions per capita in the developed world [26]. Solar assisted air conditioning provides a significant potential to reduce the consumption of primary energy, which leads to a reduction in CO_2 emissions. The environmental analysis of the Central Queensland University desiccant cooling system is based in terms of the amount of CO_2 emissions avoided by saving electrical energy used by air conditioning which given by Eq. (11).

$$CO_2\ (\text{avoided}) = CO_2\ \text{factor} \times E_S \tag{11}$$

Where CO_2 factor is the emission factor and E_S is the saved energy. As known, the main source of Australia's electricity generation is black coal. Coal-fired power plants generated 77.2% of the country's total electricity production [27].

5. Building and system parameters

In this paper, Building 41, the Health and Safety Office at Central Queensland University, Rockhmapton, campus has been investigated in terms of cooling load and energy requirements. Rockhampton's climate is sub-tropical which is characterised by long, hot and humid summers as shown in Fig. 3 [28]. Moreover the city of Rockhampton is located at −23.4°N (latitude) and 150.5°E (longitude). The city's cooling season is from November to March and winter is from June to August. Throughout the summer Rockhampton's mean maximum temperature ranges between 28 °C and 36 °C while in winter the mean maximum temperature ranges between 22 °C and 28 °C [29]. Moreover Rockhampton recorded a maximum relative humidity of 92% during the month of February while the average monthly relative humidity recorded was in the same month was 72% [26,29]. Indoor humidity is a major contributor to indoor air quality. During summer when the ambient temperature and humidity are high, dehumidifiers must be in use to dehumidify warm and moist outdoor air in order to maintain buildings humidity levels between 50% and 60% as in ASHRAE Standard 55 [30].

Building cooling load depends on many parameters like building orientation, building size, gain, ventilation, infiltration and climatic conditions [31]. The reference building cooling load is evaluated using TRNSYS 16 software. TRNSYS is a transient system simulation programme developed at the University of Wisconsin, to assess the performance of thermal and electrical energy systems [32]. The software consists of sub-routines that represent system components, which are called Types, each Type working as a module in the system [33]. Additionally the building shown in Fig. 4, is modelled as a single zone using Type56a as in Fig. 5. The buildings total area is 128 m^2 and it is 3 m in height.

The building has 10% glazing fraction for the north side, 25% for the east side, 25% for the west side and 50% for the south side.

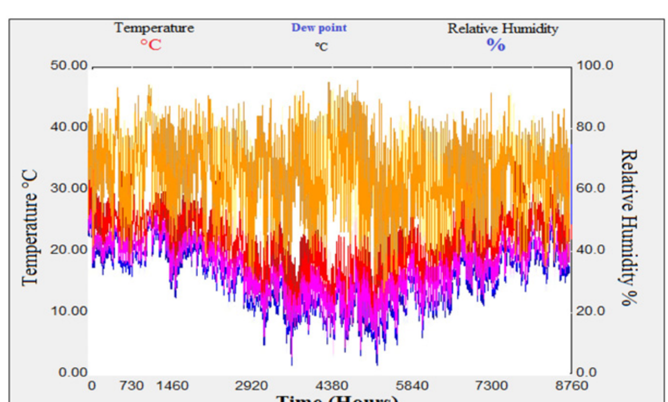

Fig. 3. Rockhampton climate data.

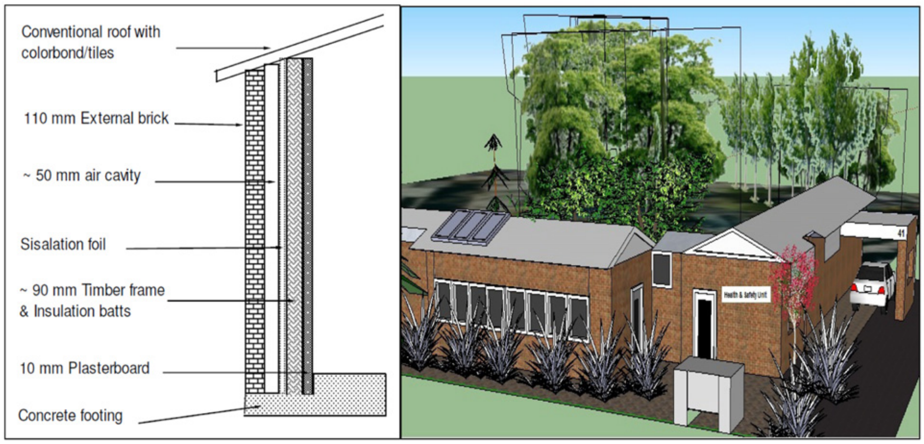

Fig. 4. Building 41 model and specifications.

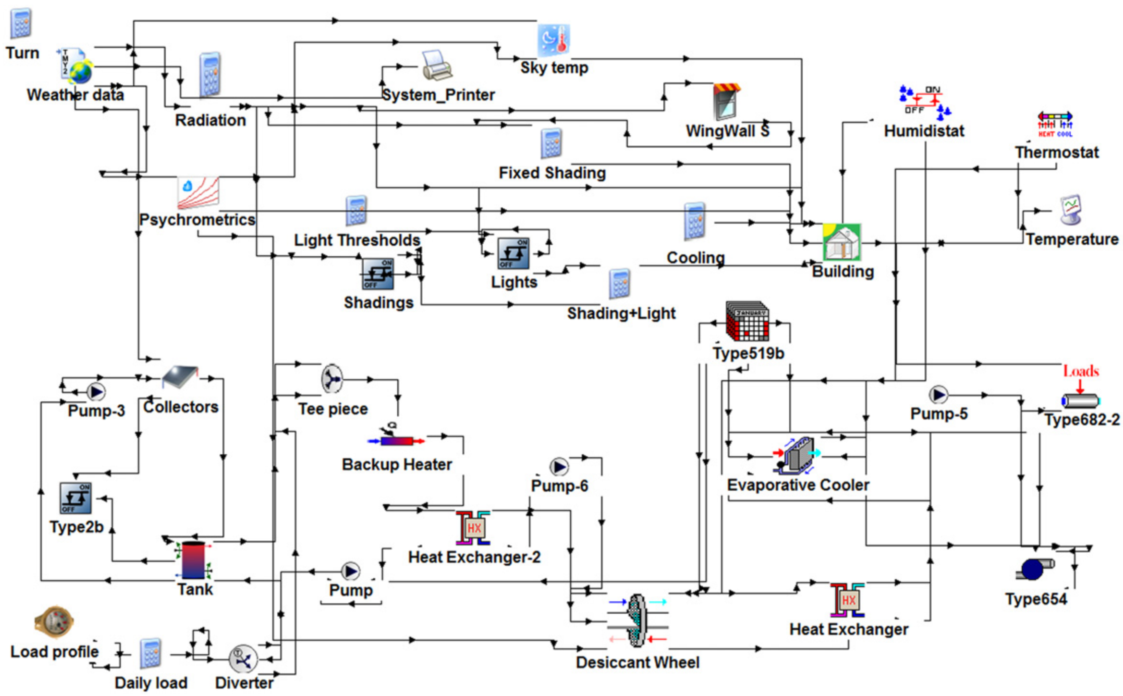

Fig. 5. Building modelling using TRNSYS 16.

The internal gains are: occupants' density is 0.1 per square metre, seven personal computers at 230 W per computer, illuminations are set to be 2 W per square metre and specific gain is 14 W per square metre. The external loads are: the fresh air flow rate of 10 l per person, infiltration rate is set to be 0.2 Vol/h at all times. The window height is 1.8 m and width is 5 m from both sides north and south, while the door is 0.9 m width and 1.8 m in length.

In order to define internal gain, default values by [34] are used. Building normal working hours of the commercial site, are from 8 am to 6 pm, Monday to Friday.

In TRNSYS 16 studio as in Fig. 5, modelling of the solar system started with Type1b which is used to model the flat plate collectors, Type4c was used to represent the hot water storage tank, Type6 was used to simulate the backup heater and Type650 was used to model the heat exchanger. Next is the desiccant wheel which was modelled as Type683 and then the evaporative cooler which was modelled as Type757a. The overall system parameters and inputs are listed in Table 1.

6. Results and discussion

6.1. Technical performance

The potential of solar desiccant cooling systems generally depends on their performance compared to conventional compression HVAC systems, in particular primary energy usage and environmental impacts. To determine the performance characteristics of the system a number of runs were performed using the model shown in Fig. 5. However, the performance of the system is reliant on various factors. The main two factors considered in this study are the collector's area and the hot water storage volume variations.

According to Fig. 6, Rockhampton's average mean irradiance of global radiation for the months of May and August was near 180 W/m^2 and for the months of June and July near 160 W/m^2. While the recorded average mean irradiance of global radiation during the cooling season was near 250 W/m^2 for the months of

Table 1
System parameters.

TRNSYS Type683 for desiccant and Type757a for indirect evaporative	
Capacity	10 kg/h
Nominal dry air flow	1450 m^3/h
External static pressure	100 Pa
Nominal wet air flow	580 m^3/h
Speed of rotor rotation	42 r/h
Process air evaporative cooler saturation efficiency	90%
Exhaust air mass flow rate	3.36 kg s^{-1}
Evaporative cooler saturation efficiency	75%
Evaporative cooler power consumption	0.1 kW
Desiccant wheel power consumption	0.2 kW
TRNSYS Type1b for FPC, Type4c for hot water storage, Type6 for backup heater, Type3 for hot water pump and Type650 for heat exchanger	
The conversion factor η_0	0.780
The lost coefficient C_1	4.2 (W/m^2 K)
The lost coefficient C_2	0.008 (W/m^2 K^2)
Fluid volume/collector area	70 l/m^2
storage tank with a loss coefficient	0.5 W/m^2 K
Backup heater with efficiency	90%
Backup heater maximum heating rate	10 kW
Hot water pump maximum power	67 W
Hot water pump flow rate	200 kg/h
Heat exchanger effectiveness	0.65

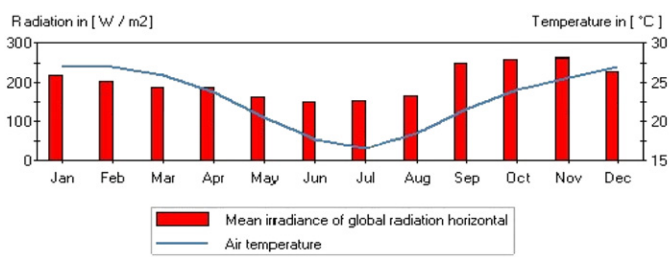

Fig. 6. Rockhampton global radiations on horizontal.

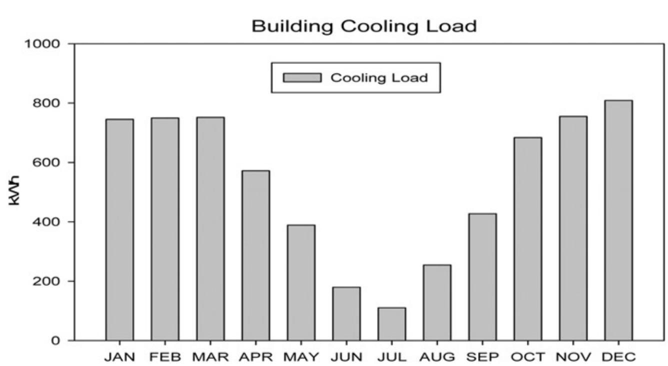

Fig. 7. Building cooling load.

September and October, near 270 W/m^2 for the month of November, near 230 W/m^2 for the months of December and January and near 200 W/m^2 for the months of March and April.

The building's total annual cooling load was 6428 kW h cooling. The maximum cooling load was in the month of December at 809 kW h cooling followed by the months of November, January, February and March as shown in Fig. 7. The minimum cooling load was in the month of July at 110 kW h cooling followed by the months of June and August (Fig. 7).

The proposed cooling system used 100% fresh air to enhance indoor air quality. Fig. 8, is the Psychrometric chart on the operation of Central Queensland University, Rockhampton desiccant cooling system. It shows that the operation started on a

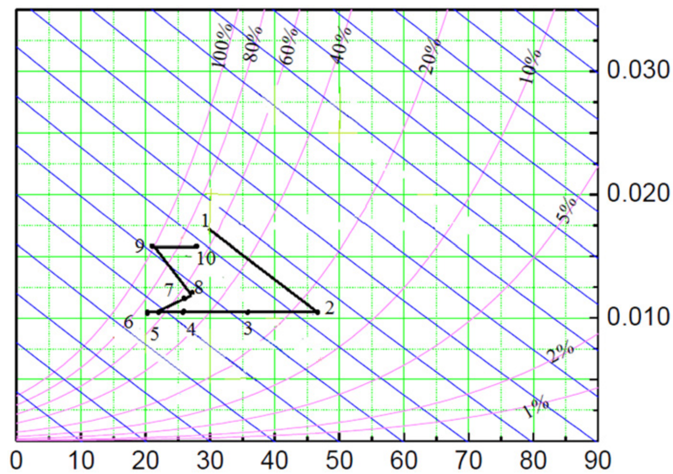

Fig. 8. Psychrometric chart on the operation of the designed system.

typical summer's day when the temperature was near 30 °C and 80% relative humidity. In stage 1 as shown in Fig. 2, the fresh air enters the desiccant wheel in order to get dried. However, drying the fresh air by the means of the desiccant wheel increased its temperature to near 45 °C and decreased its humidity to near 30% as in stage 2. Next, to decrease the air sensible temperature to near ambient level, the air will pass through an 80% efficiency heat exchanger reaching stage 3. The resultant air temperature will be near 33 °C with 30% humidity. In stage 4 the treated air temperature dropped further to near 27 °C and 50% humidity. Afterwards, the air passed to stage 5 throughout a conventional AC unit which dropped the temperature from 27 °C to near 25 °C with near 60% humidity.

The efficiency of the system can be quantified by using system coefficient of performance *COP*. Fig. 9 shows the effect of the collector's area on the coefficient of performance for the proposed cooling system. These results show the similarities of the maximum coefficient of performance (*COP*) achieved throughout the year by the desiccant cooling system/when installing 50 m^2 and 20 m^2 of solar collector's area. The maximum *COP* was 1.2, achieved in the months of January, February, October, November and December. The average *COP* achieved by the system during Rockhampton cooling period was 0.99 when installing 20 m^2 and 1.01 when installing 50 m^2. As already noted the system *COP* barely changed after installing more than 20 m^2 of solar collectors. Therefore, the recommended solar collector's area to be installed is 20 m^2.

When evaluating the system in terms of primary energy savings (solar fraction), changing the solar collector's area had a big influence. Fig. 10, shows that the solar system achieves an average of 70% of solar fraction during the cooling seasons. This can be achieved by installing 50 m^2 of solar collectors. The system solar fraction peaked in the months of December and January reaching 83%, followed by November at 79% and decreases in the month of February to 65% followed by March at 43%. Additionally, when installing 20 m^2 of solar collectors the systems best performance was in the months of December and January reaching 65%. Whilst by installing 10 m^2 and 5 m^2 of solar collectors the system best performance was recorded at 46% and 30% respectively in the months of December and January. Finally, the results showed that the annual solar fraction of the university installed system which consists of 10 m^2 of solar collector's area and 0.4 m of hot water storage is 22%.

The systems solar fraction has a direct proportion with the solar collector's total area. The performance of the solar system has been evaluated in terms of solar fraction as shown in Fig. 11.

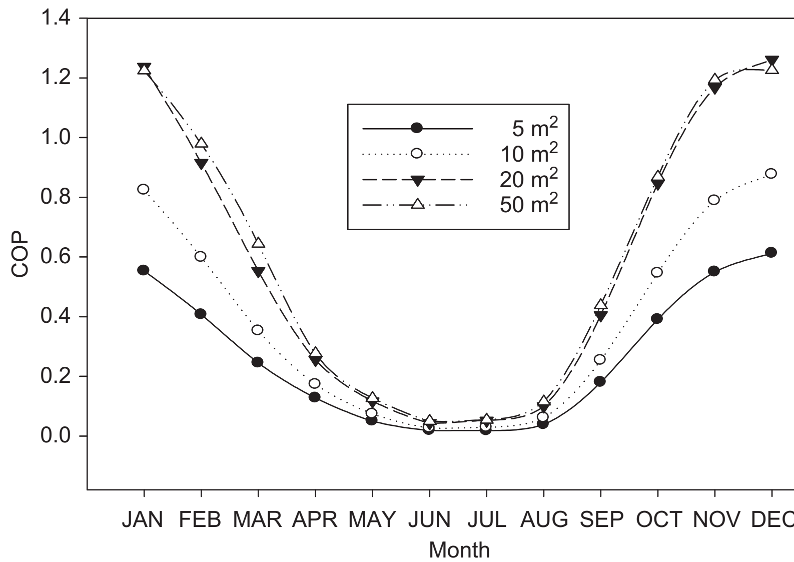

Fig. 9. The effect of changing collector's area on *COP*.

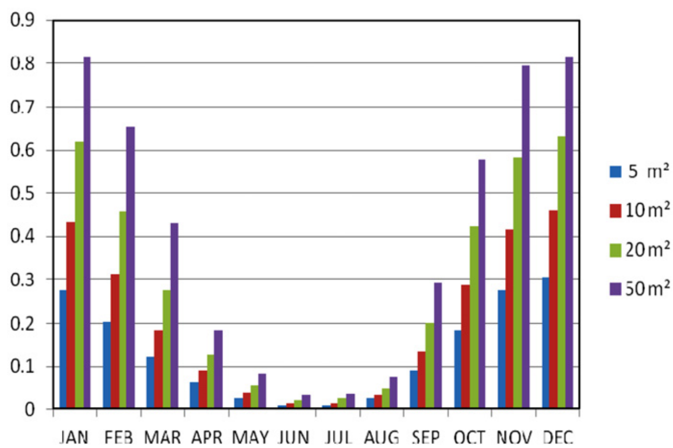

Fig. 10. Effects of solar collectors area on solar fraction.

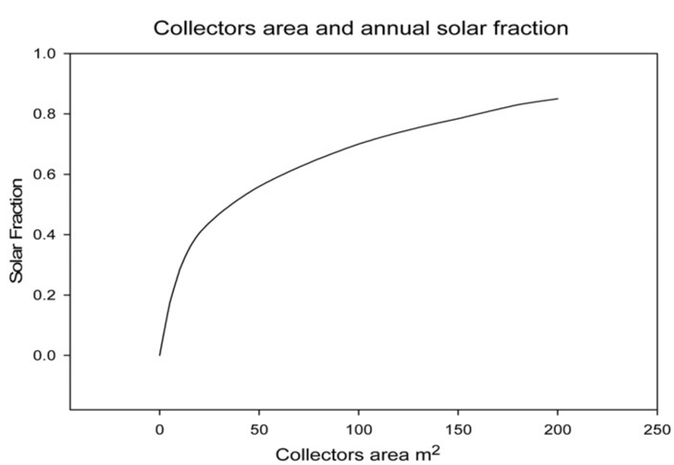

Fig. 11. Annual solar fraction.

The horizontal axis represents the collector's area and the vertical axis represents the solar fraction. There is a good power correlation between the solar fraction and collector's area, where the correlation coefficient is 0.9969. The regression can be

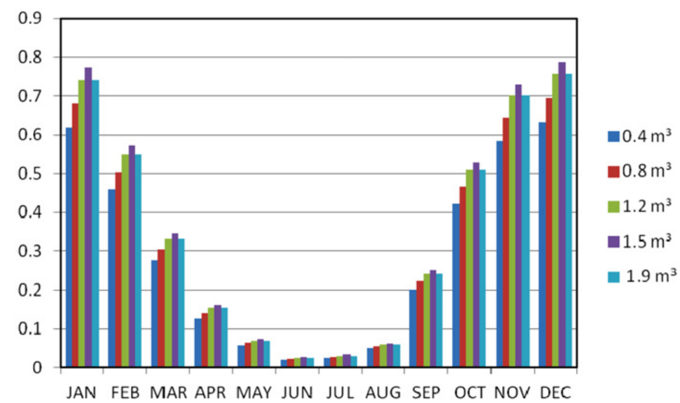

Fig. 12. Effects of hot water storage volume variation on solar fraction.

represented by a power function based on the least square method as in Eq. (12).

$$F = -0.0512 + 0.1613\,(A_C)^{0.3285} \tag{12}$$

where F is the solar fraction and A_C is the collector's total area. The relationship of the optimum solar collector's area and annual solar fraction is given by Eq. (13).

$$\frac{\partial F}{\partial A_C} = \frac{P_2 C_A}{P_1 C_F L} = 0.1613 A_C^{-0.6715} \tag{13}$$

In this study the effect of the hot water storage tank volume variation on the performance of the solar system is examined for the fixed collector's area which was 20 m^2.

Fig. 12, shows that the system annual solar fraction increased by increasing the hot water storage volume to a certain level which was 1.5 m^3. The maximum solar fraction happened in the month of December and January reaching 78%. The systems annual solar fraction was 57% when the storage volume was 0.4 m^3 and increased to the maximum of 65% annually when the storage volume was 1.5 m^3. However when increasing the storage volume from 1.5 m^3 to 1.9 m^3 the solar fraction started to decrease reaching a maximum of 73% for the months of December and January. Therefore, the recommended hot water storage volume to be installed is 1.5 m^3.

6.2. Economic performance

Central Queensland University's solar system cost was AU$ 600/m^2 while the total installed area was 10 m^2 at total cost of AU$ 6500 including all trades installations requirements. The cost of the desiccant cooling system was AU$ 15,000 installed and the cost of the evaporative cooler was AU$ 2000. The project lifetime is 25 years. The inflation rate of fuel prices in Australia varies from 2% to 8% and the discount rate is 8%. The system total cooling capacity (conventional and desiccant) is 5 kW with a 1.2 coefficients of performance (*COP*).

Fig. 13, shows that the cumulative solar savings during the operating time of 22 years. The time needed for the cash flow to reach zero was 12 years, while the time needed for the cumulative fuel solar savings to be equal to the total investment (payback time) was 22 years if the system resale value at the end of the operating time was 20% of the total cost.

In addition, Fig. 14 represents the present worth factor for system cost at the end of its operating time assuming a change will occur in fuel inflation rate and in the discount rate. However, the results show that the change in the money value after 20 years would be rapid if the inflation rate and the discount rate exceeded 10% reaching a maximum of AU$ 80,000. Keeping the inflation rate below 5% will keep the monetary value between AU$ 40,000 and AU$ 60,000.

6.3. Environmental performance

According to the Australian government's office of climate change, the CO_2 factor is 88.2 g/kW h. The annual carbon dioxide emission is considered as one of the most important indicators of environmental assessment of energy system [35].

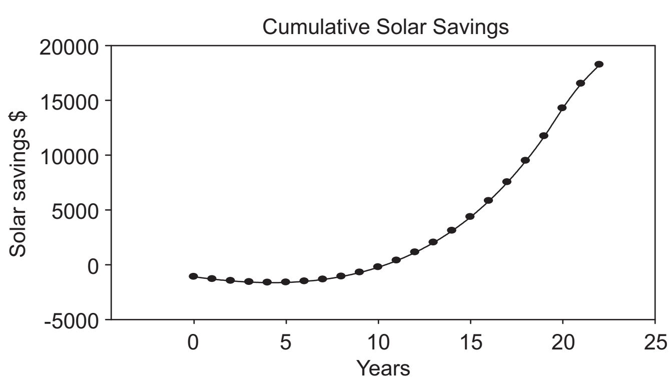

Fig. 13. System cumulative savings.

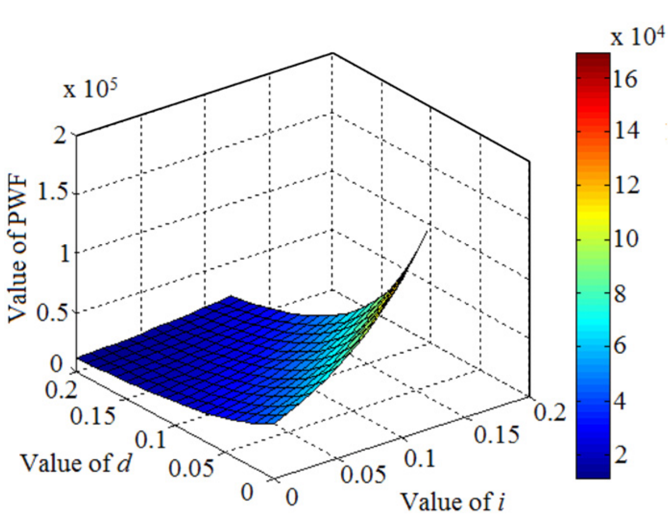

Fig. 14. System present worth factor.

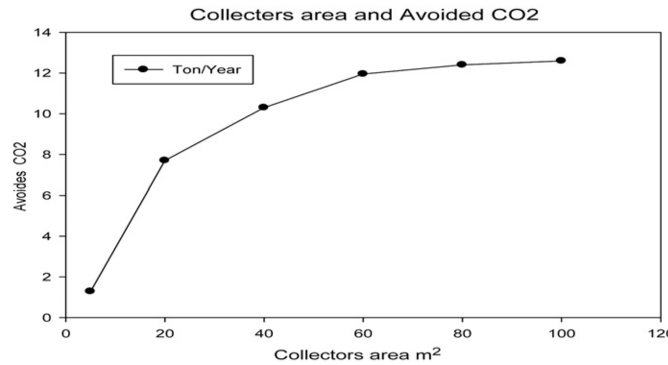

Fig. 15. Annual avoided gas emissions per m^2 of solar collectors.

It is clear that the carbon dioxide emission has no relationship to the system cost scenarios, it depends on the solar irradiation and the energy harvested from the sun. Fig. 15, shows that by installing 10 m^2 of solar collectors, the annual avoided gas emission will be nearly 4.4 tonnes of CO_2. This avoided gas emission can be enhanced by installing more solar collectors.

7. Conclusion

In this study, a solar desiccant cooling system in an institutional building was evaluated using TRNSYS 16 simulation software. The study outlined the potential of using a solar desiccant cooling system under Queensland, Australia's subtropical climate. The proposed system consists of a desiccant wheel, thermal solar system, heat exchangers, fans and an evaporative cooler. A TRNSYS model was developed using a typical meteorological year. The system's technical, economic and environmental performances were evaluated. Furthermore, this study investigated the factors that affected the performance of a solar desiccant cooling system like different solar collector's areas and hot water storage volume variation. The results showed that the total annual cooling load was 6428 kWh. It reached its peak in the month of December recording a maximum of 809 kWh and the minimum was in the month of July at 110 kWh. The Central Queensland University cooling system, which consists of 10 m^2 of collector's area, achieved a 22% of annual solar fraction of total required energy and a 0.7 *COP*. The system performance can be boosted to 60% of energy savings and a 1.2 *COP*. In addition the economical analysis showed that the system will cover its total installation cost in 22 years if the system materials and equipments resale value reached a minimum of 20%. The environmental analysis showed that the system is able to avoid 4.4 tonnes of green house gas emission (CO_2).

Despite the fact that solar assisted air conditioning systems are characterised by high installation costs and the lack of technical information between developers and decision makers they can still deliver a comfort level at a small energy cost and less gas emissions. Furthermore system reliability will increase significantly if solar cooling equipments prices decrease and their efficiencies increase.

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